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(71) Applicant: NIPPONDENSO CO., LTD.
Kariya-city Aichi-pref., 448 (JP)

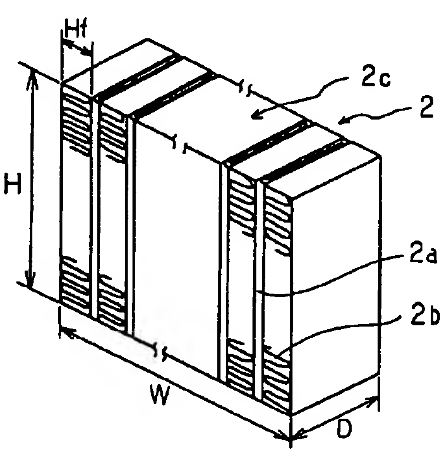
(72) Inventors:
- Fukuoka, Mikio
Kariya-city, Aichi-pref. (JP)
- Aki, Yoshifumi
Kariya-city, Aichi-pref. (JP)

(74) Representative: Klingseisen, Franz, Dipl.-Ing. et al
Patentanwälte,
Dr. F. Zumstein,
Dipl.-Ing. F. Klingseisen,
Postfach 10 15 61
D-80089 München (DE)

# (54) Corrugate fin type heat exchanger

(57) According to the present invention, in a corrugate fin type heat exchanger (2) including a core portion (2c) having a plurality of flat tubes (2a) disposed in parallel with flow direction of the air and at least one corrugate fin (2b) disposed between each pair of the flat tubes (2a), an inner thickness of the flat tube (2a) is in a range of 0.6 - 1.2 mm, a height of the corrugate fin (2b) is in a range of 3 - 3 mm, and a ratio (St/WxD) of the cross-sectional area (WxD) expressed by an overall width dimension (W) and a thickness dimension (D) of the core portion (2c) to a total cross-sectional flow passage area (St) of the plurality of flat tubes (2a) is set to a range of 0.07 - 0.24 according to the inner thickness of the flat tube (2a) and the height of the corrugate fin (2b). In this way, it is possible to reduce the Reynold's number of the flow passages within the flat tubes (2a) to keep a laminar region constantly irrespective of the variation in the hot water flow quantity, thereby reducing the variation in the water side heat transfer rate.

FIG. 3

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## Description

### BACKGROUND OF THE INVENTION

1. Field of the invention:

The present invention generally relates to a corrugated fin type heat exchanger for heating air by heat exchanging hot water with the air, and is preferably applied to a corrugate fin type heat exchanger for heating used in an automotive air conditioner in which hot water flow quantity widely varies.

2. Related Art:

In a vehicle, as illustrated in FIG. 1, a heat exchanger 2 for heating is installed in a cooling water (hot water) circuit of an engine 1 for running the vehicle. Hot water is circulated into the heat exchanger 2 by a water pump 3 driven by the engine 1, and the flow quantity of the hot water flowing from a flow quantity control valve 4 into the heat exchange 2 is controlled to adjust the temperature of the air flow of the heat exchanger 2.

Engine cooling water is circulated into a radiator 6 by the water pump 3 through a thermostat 5 to cool the engine cooling water within the radiator 6. The thermostat is a well-known device, in which a valve opens when the cooling water temperature rises to or exceeds a predetermined temperature, thereby the cooling water flowing into the radiator 6.

The reference numeral 7 denotes a bypass circuit for the engine cooling water, 8 denotes a radiator side circuit, and 9 denotes a heater side circuit. The water pump 3 circulates the cooling water through all these circuits 7, 8 and 9.

However, as the water pump 3 is driven by the engine 1, a rotational speed of the water pump 3 largely varies according to the rotational speed of the engine 1, i.e., the vehicle speed, and thereby flow quantity of the hot water into the heat exchanger 2 largely varies.

As a result of such large variation in the flow quantity of the hot water into the exchanger 2, when the vehicle is running at a low speed (when the hot water flow quantity is small), as illustrated in FIG. 2, there is a problem in that the heat radiation performance of the heat exchanger 2 is extremely deteriorated.

In FIG. 2, the ordinate represents the heat radiation performance Q of the heat exchanger 2, the abscissa represents the flow quantity Vw of the hot water into the heat exchanger 2. As can be seen from Fig. 2, the hot water flow quantity is 16 lit/min when the vehicle is running at 60 km/h, and the hot water flow quantity is 4 lit/min when the vehicle is in idling. As the hot water flow quantity decreases, the heat radiation performance when the vehicle is in idling falls by 22 % down as compared to when the vehicle is running at 60 km/h. As a result, heating feeling is deteriorated.

Particularly when the vehicle is running in urban streets, as the vehicle is subjected to frequent starts and stops due to traffic signals. Therefore, there is a problem in that whenever the vehicle comes to be in idling, the passenger feels insufficient in heating, and heating feeling is excessively deteriorated.

The inventors of the present invention have studied the cause of such deterioration of the heat radiation performance from various points of view and found out the following reasons.

As illustrated in FIG. 3, the heat exchanger 2 includes a plurality of flat tubes 2a arranged in parallel with the air flow direction. These flat tubes 2a are individually disposed in a single row in the air flow direction. Corrugate fins 2b are disposed between each pair of flat tubes 2a, thereby configuring a corrugate type heat exchanger. The reference numeral 2c denotes a core portion which is composed of the flat tubes 2a and the corrugate fins 2b.

In FIG. 4, the ordinate represents water side heat transfer rate $\alpha w$ of the flat tube 2a, and the abscissa represents the Reynold's number Re and hot water flow quantity Vw of the hot water passages formed with the flat tubes 2a.

As understood from FIG. 4, the Reynold's number is within a range of 500 - 2200 when the hot water flowing into the heat exchanger 2 is within a predetermined range (16 lit/min when the vehicle is running at 60 km/h, and 4 lit/min when the vehicle is in idling), and the heat exchanger 2 is operated to the extent from the laminar region to a transition flow region. For this reason, the water side heat transfer rate $\alpha w$ largely varies in accordance with the variation of the hot water flow quantity. As a result, it turned out that the water side heat transfer rate $\alpha w$ largely falls within the low flow quantity region, thereby causing the deterioration of the heat radiation performance when the vehicle is in idling.

FIG. 4 illustrates the results of an experiment in which normal tubes with no dimples (concave and convex portion) for facilitating the turbulence of the hot water on the inner surfaces were used as the flat tubes 2a.

For improving the water side heat transfer rate $\alpha w$, in general, the turbulence of the hot water within the tubes is often facilitated. Concretely, it has been proposed that a turbulence generator for facilitating turbulence is inserted into the tubes or dimples for facilitating turbulence is formed on the inner surfaces of the tubes.

Therefore, the inventors of the present invention have measured the water side heat transfer rate $\alpha w$ by using the flat tubes 2a with dimples for facilitating turbulence. As a result, as illustrated in FIG. 5, the flat tube with dimples could generally improve the water side heat transfer rate $\alpha w$ as compared to the normal tube, and the Reynold's number Re of the dimple tube in the transition region from laminar to turbulence decreased from 1400 with the normal tube to 1000.

However, the large variation in the water side heat transfer rate $\alpha w$ according to the hot water flow quantity still remained even when the flat tube with dimples is used. Therefore, even when a technique for facilitating a turbulence such as the flat tubes with dimples is used, it

is not possible to solve the problem in that the heat radiation performance when the hot water flow quantity is small (when the vehicle is running at a low speed) is deteriorated.

## SUMMARY OF THE INVENTION

In view of the above problems, the present invention has an object to provide a corrugate type heat exchanger which can effectively improve the heat radiation performance within a low flow quantity region.

As understood form FIGS. 4 and 5, when the Reynold's number of approximately 1000 was taken as a transition point, the variation (inclination) of the water side heat transfer rate $\alpha w$ against the Reynold's number within the laminar region was very small in the region with the Reynold's number of 1000 or less.

In consideration of such small variation (inclination) of the water side heat transfer rate $\alpha w$ within the laminar region, in the present invention, the Reynold's number of the flow passages of the flat tubes is set to be extremely small to keep water flow in the flow passages of the flat tubes in a complete laminar region over the regular use range of the hot water flow quantity from the high flow quantity region to the low flow quantity region, thereby reducing the variation in the water side heat transfer rate $\alpha w$ and increasing the water side heat transfer rate $\alpha w$ simultaneously to improve the heat radiation performance within the low flow quantity region.

According to the present invention, in a corrugate fin type heat exchanger including a core portion having a plurality of flat tubes disposed in parallel with flow direction of the air and at least one corrugate fin disposed between each pair of the flat tubes, an inner thickness of the flat tube is in a range of 0.6 - 1.2 mm, a height of the corrugate fin is in a range of 3 - 6 mm, and a ratio (St/W×D) of the cross-sectional area (W×D) expressed by an overall width dimension (W) and a thickness dimension (D) of the core portion to a total cross-sectional flow passage area (St) of the plurality of flat tubes is set to a range of 0.07 - 0.24 according to the inner thickness of the flat tube and the height of the corrugate fin.

It is preferable that the Reynold's number being set to 1000 or less when flow quantity of the hot water passing through the core portion is 16 lit/min.

Further, it is preferable that the flat tubes and the corrugate fins being made of aluminum, a wall thickness of the flat tube being set to a range of 0.2 - 0.4 mm, and a wall thickness of the corrugate fin being set to a range of 0.04 - 0.08 mm.

According to the present invention as the above, it is possible to reduce the Reynold's number of the flow passages of the flat tubes and to keep the laminar region constantly even if the hot water flow quantity widely varies. As a result, the variation in the water side heat transfer rate can be reduced.

Furthermore, it is possible to improve the water side heat transfer rate sufficiently by setting the inner thickness of the flat tube to a thin width dimension in a range of 0.6 - 1.2 mm, and to improve the heat radiation performance by setting the height (Hf) of the corrugate fin to the optimum range in a range of 3 - 6 mm.

As a result, even in the low flow quantity region of the hot water flow quantity, tit is possible to largely improve the heat radiation performance cas compared to the conventional type, thereby heating feeling for the user of the heating system being remarkably improved.

Particularly in an automotive air conditioning system, since the hot water flow quantity frequently varies due to the repetition of starts and stops of a vehicle, the improvement in the heating feeling as described above is extremely useful.

## BRIEF DESCRIPTION OF THE DRAWINGS

Additional objects and advantages of the present invention will be more readily apparent from the following detailed description of preferred embodiments thereof when taken together with the accompanying drawings in which:

FIG. 1 is a diagram illustrating an engine cooling water circuit;
FIG. 2 is a graph illustrating the relationship between the hot water flow quantity and heat radiation performance of the conventional heat exchanger;
FIG. 3 is a perspective view illustrating the core portion of a heat exchanger of an embodiment according to the present invention;
FIG. 4 is a graph illustrating the relationship among the hot water flow quantity, Reynold's number and water side heat transfer rate of the conventional heat exchanger;
FIG. 5 is a graph illustrating the relationship among the hot water flow quantity, Reynold's number and water side heat transfer rate of another conventional heat exchanger;
FIG. 6 is a graph illustrating the relationship between the corrugate fin height and heat radiation performance of an the heat exchanger of the embodiment according to the present invention;
FIG. 7 is a graph illustrating the relationship between the total cross-sectional area ratio of flat tubes and Reynold's number of the heat exchanger of the embodiment according to the present invention;
FIG. 8 is a cross-sectional view illustrating the flat tube of the heat exchanger of the embodiment according to the present invention;
FIG. 9 is a graph illustrating the relationship between the hot water flow quantity and heat radiation performance of the heat exchanger of the embodiment according to the present invention;
FIG. 10A is a graph illustrating the relationship between the inner thickness of flat tube and heat radiation performance of the heat exchanger of the embodiment according to the present invention;

FIG. 10B is a graph illustrating the relationship between the inner thickness of the flat tube and water side heat transfer rate of the heat exchanger of the embodiment according to the present invention;

FIG. 11 is a graph illustrating the relationship among the total cross-sectional area ratio of flat tubes Reynold's number and corrugate fin height of the heat exchanger of the embodiment according to the present invention;

FIG. 12 is a graph illustrating the relationship among the total cross-sectional area ratio of flat tubes, inner thickness of flat tube and corrugate fin height of the heat exchanger according to the present invention;

FIG. 13 is a graph illustrating the relationship between the hot water flow quantity and heat radiation performance of the heat exchanger of the embodiment according to the present invention;

FIG. 14 is a graph illustrating the relationship among the hot water flow quantity, Reynold's number and water side heat transfer of the heat exchanger of the embodiment according to the present invention as compared to the conventional type;

FIG. 15 is a half cross-sectional front view illustrating an embodiment of the heat exchanger according to the present invention; and

FIGS. 16A-16F are schematic front views illustrating modifications of the heat exchanger according to the present invention.

## DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

An embodiment of the present invention will now be described with reference to the drawings.

In FIG. 3, dimensions W (width), D (thickness) and H (height) of the core portion 2c of the heat exchanger 2 are generally set as W = 100 - 300 mm, D = 16 - 42 mm and H = 100 - 300 mm in consideration of mounting the heat exchanger 2 easily within a heater unit housing of an automotive air conditioning system and the required heat radiation performance.

As illustrated in FIG. 6, it is optimized that the height Hf of a corrugate fin 2b is set in a range of 3 - 6 mm with 4.5 mm in the center, in consideration of the heat radiation performance, which is described in the Japanese Unexamined Patent Publication No. 5-196383, the content is incorporated herein by reference.

To keep the flow passages within flat tubes 2a as a laminar region constantly by setting the Reynold's number Re to a small value, the flow velocity v of hot water within the flat tubes 2a and the equivalent diameter de of the flat tube 2a should be reduced by using the following equation (1).

$$Re = v \cdot de/\nu \qquad (1)$$

where $\nu$ is the kinematic viscosity of the hot water within the flat tubes 2a, and the substantial round-hole diameter de of the flat tube 2a is the diameter of the round-hole having the same area as the cross-sectional area of the flat tube 2a.

To reduce the flow velocity v of the hot water within the flat tubes 2a, the total area St of the flow passages of the flat tubes 2a should be increased by using the following equation (2).

$$v = Vw/St \qquad (2)$$

where Vw is the flow quantity of the hot water flowing into the heat exchanger 2 and St is the sum total of the cross-sectional areas of the flow passages within all the flat tubes 2a of the core portion 2c.

To reduce the substantial diameter de of the flat tube 2a, the cross-sectional area A of the flow passage per flat tube 2a should be reduced by using the following equation 3.

$$de = 4 \cdot A/L \qquad (3)$$

where L is the wet edge length within the flat tube 2a (the length of the inner peripheral wall of the cross-sectional shape of the flat tube 2a, which will be described later with reference to FIGS. 7 and 8).

A liquid mixture of an anti freeze solution containing a rust preventive and water combined at approximately 50:50 is generally used for the hot water (engine cooling water) circulating into the heat exchanger 2, and the hot water temperature is maintained to approximately 85° C by the thermostat 5.

Here, to reduce the cross-sectional flow passage area A per flat tube 2a and to increase the total cross-sectional flow passage area St of the flat tubes 2a are contrary to each other. Therefore, to increase the total cross-sectional tube area St while reducing the cross-sectional flow passage area A per flat tube 2a, it is preferable that the core portion 2c of the following construction being employed.

The core portion 2c should be an one way flow type (full-pass type) having the cross-sectional area (W×D) of the core portion 2c in which the hot water flows only in one direction instead of U-turn type in which the hot water flows in a U-turn, and the number of the flat tubes 2a having the cross-sectional area (W×D) of the core portion 2c, through which the hot water flows in parallel, should be increased. The concrete structure of the core portion 2c of the one way flow type (full-pass type) will be described later with reference to FIG. 15.

Next, for the core portion 2c dimensioned to W (width) = 180 mm, H (height) = 180 mm and D (thickness) = 27 mm, the inventor of the present invention examined the total cross-sectional flow passage area St of the flat tubes 2a which could hold the Reynold's number Re to be 1000 or less (within the complete laminar region in FIG. 5) until the hot water flow quantity Vw increases to 16 lit/min, which is a flow quantity when the vehicle is running at a speed of 60 km/h.

Since the total cross-sectional flow passage area St of the flat tubes 2a varies according to the size (W, D) of the core portion 2c, the inventors examined the relationship between the ratio (St/WxD) of the total cross-sectional flow passage area St of the flat tubes 2a to the cross-sectional area of the core portion 2c (WxD) and the Reynold's number Re as a parameter of the inner thickness b of the flat tube 2a within a range of 0.5 - 1.7, as illustrated in Fig. 7. In Fig. 7, the abscissa represents the ratio (St/WxD) and the ordinate represent the Reynold's number Re.

The inner thickness "b" of the flat tube 2a means the thickness in the short side direction of the flow passage within the flat tube 2a in the cross-sectional shape of the flat tube 2a illustrated in FIG. 8, and the width dimension of the long side direction is indicated with "a".

In the experiment which result is illustrated in FIG. 7, the inner width "a" of the flat tube 2a was fixed to 26.5 mm and the inner thickness "b" was changed.

The ratio (St/WxD) with respect to each thickness "b" of the flat tube 2a, where the Reynold's number Re is 1000, is indicated with ○. As illustrated in FIG. 7, the ratio (St/WxD) with respect to each thickness "b" of the flat tube 2a where the Reynold's number Re is 1000 or less exists in a large number.

Therefore, the inventors of the present invention also studied the optimum thickness "b" of the flat tube 2a in view of its performance, and further studied the relationship between the optimum thickness "b" and the total cross-sectional flow passage area St of the flat tubes 2a.

Specifically, the inventors studied on the core portion 2c with the width W = 180 mm, the height H = 180 mm and the thickness D = 27 mm, and the fin height Hf being the central value 4.5 mm of the optimum range (3 - 6 mm) to optimize the thickness "b" of the flat tube 2a in view of its performance.

In FIG. 9, the ordinate represents the heat radiation performance Q of the heat exchanger 2 and the abscissa represents the flow quantity Vw of the hot water circulating into the heat exchanger 2. The heat radiation performance Qo with the hot water flow quantity Vwo determined according to the matching point of the water flow resistance of the heat exchanger 2 and the pump characteristics of a water pump 3 of an engine 1 corresponds to the performance of the heat exchanger 2 in an actual operation.

The heat radiability Qo of the heat exchanger 2 in an actual operation is obtained by varying the thickness "b" of the flat tube 2a and summarized in Fig. 10A. In Fig. 10A, the heat radiation performance Qo of the thickness b = 0.7 mm at which the heat radiation performance Qo of the heat exchanger 2 in an actual operation is the highest is set to 100, the ordinate represents the percentage of the heat radiation performance Qo of each thickness "b" of the flat tube 2a against the heat radiation performance Qo = 100 of such thickness b = 0.7 mm of the flat tube 2a.

It is understood from FIG. 10A that the optimum range of the thickness "b" of the flat tube 2a is 0.6 - 1.2 mm.

FIG. 10B illustrates the relationship between the thickness b of the flat tube 2a and water side heat transfer rate αw with the Reynold's number Re = 500. The smaller the dimension "b" is, the higher the water side heat transfer rate αw is. As a matter of fact, however, when the dimension "b" decreases, the inner resistance of the flat tube 2a increases. Resultantly, the flow quantity of the circulating hot water decreases, and the heat radiation performance is deteriorated, as illustrated in FIG. 10A. Therefore, it is necessary to set the lower limit of the thickness "b" to 0.6 mm.

Based on the above results, the optimum range of the ratio of the total cross-sectional flow passage area of the flat tube 2a (St/WxD) is obtained from the optimum range of the fin height Hf (3 - 6 mm) and the optimum range of the thickness b (0.6 - 1.2 mm). The shaded portion X in FIG. 11 indicates the optimum range.

As illustrated in FIG. 12, when this optimum range is rewritten by taking the total cross-sectional flow passage area ratio (St/WxD) of the flat tubes 2a as the ordinate and the thickness "b" of the flat tube 2a as the abscissa, in a combination of the optimum fin height (Hf = 3 - 6 mm) and the optimum tube thickness (b = 0.6 - 1.2 mm), the total cross-sectional flow passage area ratio (St/WxD) of the flat tubes 2a is identical to the shaded portion enclosed with A, B, C and D in FIG. 12, i.e., the range of 0.07 - 0.24.

By setting total cross-sectional flow passage area ratio (St/WxD) of the flat tubes 2a within the shaded portion enclosed with A, B, C and D, it is possible to control the Reynold's number Re of the flow passage of the flat tube 2a to 1000 or less within the range of hot water flow quantity for the heat exchanger 2 (maximum 16 lit/min), thereby keeping the hot water flow within the flow passage of the flat tube 2a laminar constantly.

Now, the heat radiation performance of the heat exchanger 2 specially designed based on the above specification range is illustrated in FIG. 13. The heat exchanger 2 illustrated in FIG. 13 is dimensioned to the width W = 180 mm, height H = 180 mm and thickness D = 27 mm in the core portion 2c, the height Hf = 4.5 mm in the corrugate fin 2b, and the thickness b = 0.9 mm in the flat tube 2a, which are the central values of the optimum range, respectively.

The total cross-sectional flow passage area ratio (St/WxD) Of the flat tube 2a is 14.5. The heat radiation performance Q of the heat exchanger 2 specially designed as the above was obtained. As a result, as illustrated in FIG. 13, the heat radiation performance Q at a low flow quantity (4 lit/min when the vehicle is in idling) decreased by as small as approximately 11% down from the heat radiation performance Q at a high flow quantity (16 lit/min when the vehicle is running at 60 km/h running), which is a half or less as much as the reduction percentage (22%) in heat radiation performance of the

conventional heat exchanger 2 illustrated in FIG. 2. As clearly understood, the performance is largely improved.

In FIG. 14, the relationship between the Reynold's number Re and water side heat transfer rate αw of the heat exchanger 2 based on the specifications defined in FIG. 13 is summarized. As understood from FIG. 14, the heat exchanger 2 according to the present invention is used within a complete laminar region with the Reynold's number Re of 1000 or less, where the hot water flow quantity is 4-16 lit/min, and furthermore, the water side heat transfer rate αw within the low flow quantity region is largely improved as compared to the conventional heat exchanger.

Next, an embodiment where the heat exchanger 2 designed based on the above specifications is applied to an automotive air conditioning system is described with reference to Fig. 15. The core portion 2c is composed of the flat tubes 2a and the corrugate fin 2b. Each flat tube 2a is supportably connected to core plated 2d at both ends. Tanks 2e and 2f are connected to the core plates 2d, respectively. Further, inlet and outlet pipes 2g and 2h are detachably connected to the tanks 2e and 2f by seal joints 2i and 2j, respectively.

In FIG. 15, for example, when the pipe 2g is connected to the hot water inlet side of the hot water circuit of the engine 1, the hot water from the hot water inlet pipe 2g flows through the hot water inlet tank 2e, the flat tubes 2a, the hot water outlet tank 2f and the hot water outlet pipe 2h in this order.

That is, an one-way flow type (full-pass type) is configured in such a manner that the hot water inlet tank 2e is disposed at an end portion of the core portion 2c over the overall width direction, the hot water outlet tank 2f is disposed at the other end portion of the core portion 2c over the overall width direction, and the hot water flows only in one direction from the inlet tank 2e to the outlet side tank 2f through the flat tube 2a.

In the heat exchanger 2 configured as the one way flow type (full-pass type), it is easily possible to decrease the cross-sectional area A per flat tube 2a and increase the total cross-sectional area St of the entire flat tubes 2a simultaneously.

The heat exchanger 2 illustrate in FIG. 15 is made of aluminum. The flat tube 2a, the core plate 2d and the tanks 2e and 2f are formed from aluminum-clad material in which the aluminum core material is clad with brazing material at one or both sides. On the other hand, the corrugate fin 2b is formed from bear aluminum material which is not clad with brazing material. The heat exchanger 2 is integrally constructed by temporarily assembling these components, heating the assemblies within a brazing furnace to a brazing temperature, and then integrally brazing the assemblies.

Here, it is preferable in view of heat transfer rate, strength, etc. that the thickness of the aluminum flat tube 2a being set to a range of 0.2 - 0.4 mm and the thickness of the aluminum corrugate fin 2b being set to a range of 0.04 - 0.08 mm.

FIGS. 16A-16F illustrates modifications of the tank portion of the heat exchanger 2. FIGS. 16A to 16C illustrate modifications in which the width of the core portion 2c is set the same as that of the tanks 2e and 2f and the positions of the hot water inlet and outlet pipes 2g and 2h are differently modified.

FIGS. 16D to 16F illustrate modifications in which each width of the tanks 2e and 2f is set larger than that of the core portion 2c and the hot water inlet and the positions of the outlet pipes 2g and 2h are differently modified.

In FIGS. 15 and 16, since the shape of the heat exchanger 2 is symmetric with respect to the hot water flow direction of the core portion 2c, the tank 2e may be disposed on the hot water outlet side and the tank 2f may be disposed on the hot water inlet side contrary to the above embodiment.

The present invention having been described should not be limited to the disclosed embodiments, but it may be modified in many other ways without departing from the scope and the spirit of the invention. Such changes and modifications are to be understood as being included with the scope of the present invention as defined by the appended claims.

## Claims

1. A corrugate fin type heat exchanger (2) for heat exchanging hot water with the air, said corrugate fin type heat exchanger (2) comprising:

    a plurality of flat tubes (2a) disposed in parallel with flow direction of said air, each flat tube (2a) disposed as a single line in said flow direction; and

    at least one corrugate fin (2b) disposed between each pair of said flat tubes (2a) and connected thereto;

    said plurality of flat tubes (2a) and said corrugate fin (2b) composing a core portion (2c),

    wherein an inner thickness of the flat tube (2a) is in a range of 0.6 - 1.2 mm;

    a height of said corrugate fin (2b) is in a range of 3 - 6 mm; and

    a ratio (St/WxD) of the cross-sectional area (WxD) expressed by an overall width dimension (W) and a thickness dimension (D) of said core portion (2c) to a total cross-sectional flow passage area (St) of said plurality of flat tubes (2a) is set to a range of 0.07 - 0.24 according to said inner thickness of said flat tube (2a) and said height of said corrugate fin (2b).

2. A corrugate fin type heat exchanger (2) according to claim 1, wherein said heat exchanger (2) is applied to an heat exchanger for heating, which is used in an automotive air conditioning system in which said hot water is circulated by a water pump (3) driven by an automotive engine, and the Reynold's number being set to 1000 or less when flow quantity of said

hot water passing through said core portion (2c) is 16 lit/min.

3. A corrugate fin type heat exchanger (2) according to claim 1,
wherein said flat tubes (2a) and said corrugate fin (2b)s are made of aluminum,
a wall thickness of said flat tube (2a) is set to a range of 0.2 - 0.4 mm; and
a wall thickness of said corrugate fin (2b) is set to a range of 0.04 - 0.08 mm.

4. A corrugate fin type heat exchanger (2) according to claim 1, further comprising:
a hot water inlet tank (2e)disposed at one end of said core portion (2c), for introducing said hot water into said flat tube (2a); and
a hot water outlet tank (2f) disposed at the other end of said core portion (2c), for receiving said hot water flowing from said flat tubes (2a),
wherein said core portion (2c) is constructed in such a manner that said hot water flows only in one direction from said hot water inlet tank (2e) to said hot water outlet tank (2f).

5. A corrugate fin type heat exchanger (2) according to claim 4, further comprising:
an inlet pipe (2g) connected to said hot water inlet tank (2e) to introduce said hot water into said inlet tank (2e);
an outlet pipe (2h) connected to said hot water outlet tank (2f) to lead said hot water out of said outlet tank (2f).

6. A corrugate fin type heat exchanger (2) according to claim 5, wherein said inlet pipe (2g) and said outlet pipe (2h) extend in a longitudinal direction of said hot water inlet tank (2e) and said hot water outlet tank (2f), respectively.

7. A corrugate fin type heat exchanger (2) according to claim 5, wherein said inlet pipe (2g) and said outlet pipe (2h) extend in a lateral direction of said hot water inlet tank (2e) and said hot water outlet tank (2f), respectively.

8. A corrugate fin type heat exchanger (2) according to claim 1, wherein said heat exchanger (2) is applied to an heat exchanger for heating, which is used in an automotive air conditioning system in which said hot water is circulated by a water pump (3) driven by an automotive engine and passes through a radiator (6) for cooling said hot water by heat exchanging with air and being disposed in a cooling water pipe (8) communicating between said engine and said radiator (6), and said heat exchanger (2) being disposed in a hot water pipe (9) arranged in parallel with said cooling water pipe (8).

FIG. 1

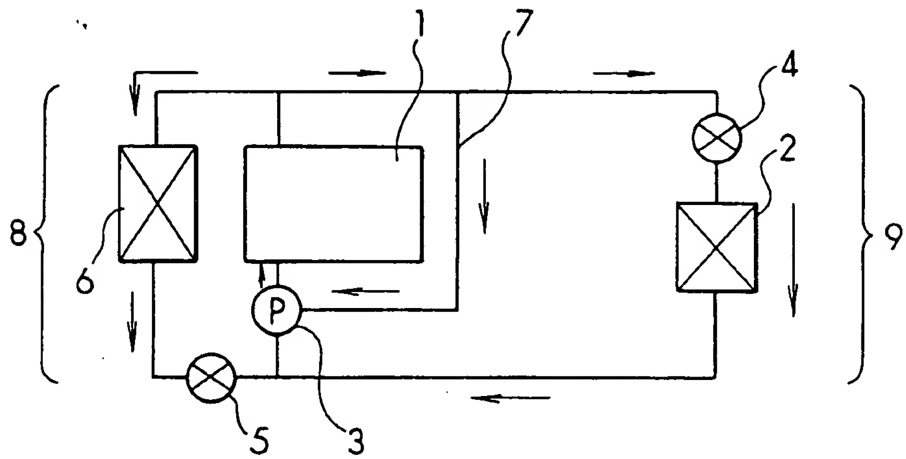



FIG. 2 PRIOR ART

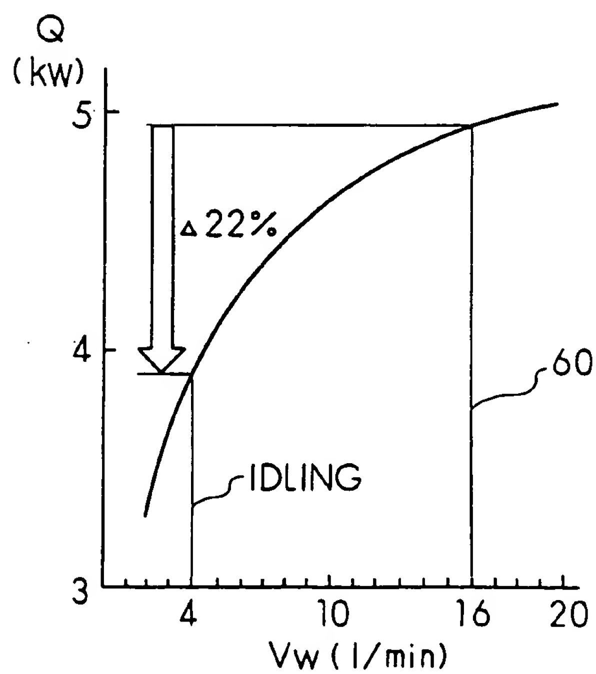


FIG. 3

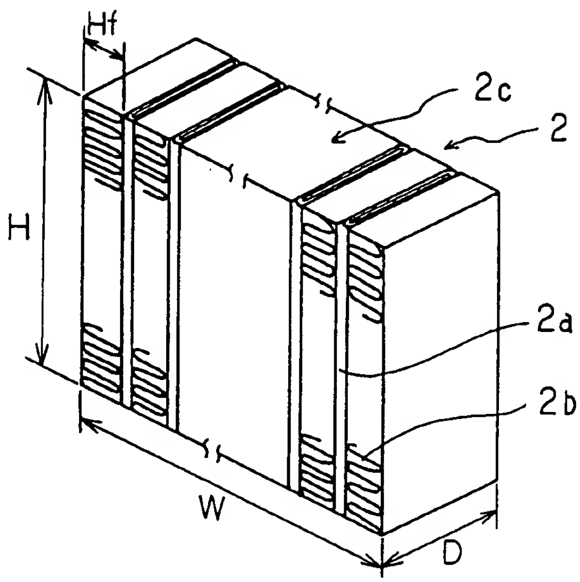



FIG. 4 PRIOR ART

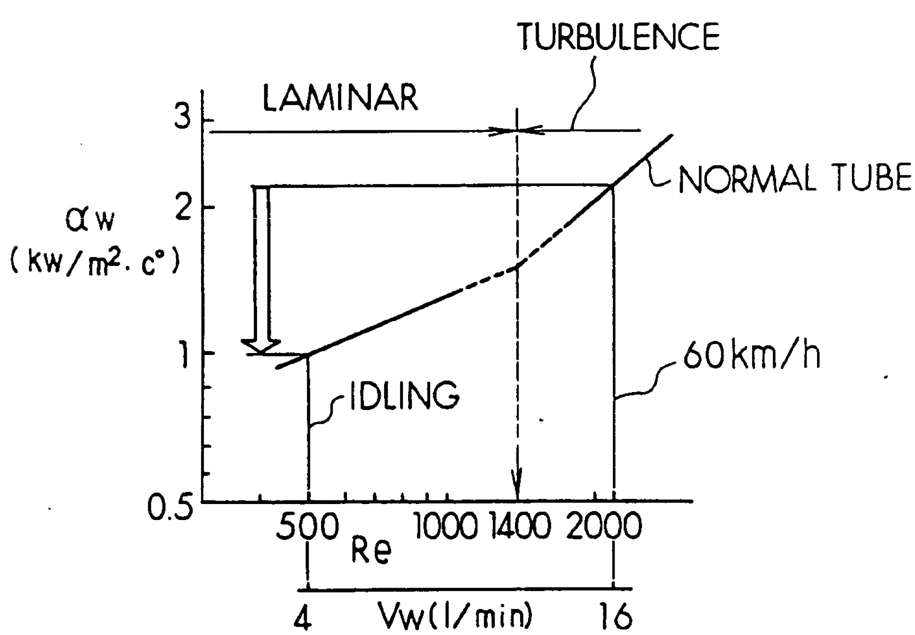


FIG. 5

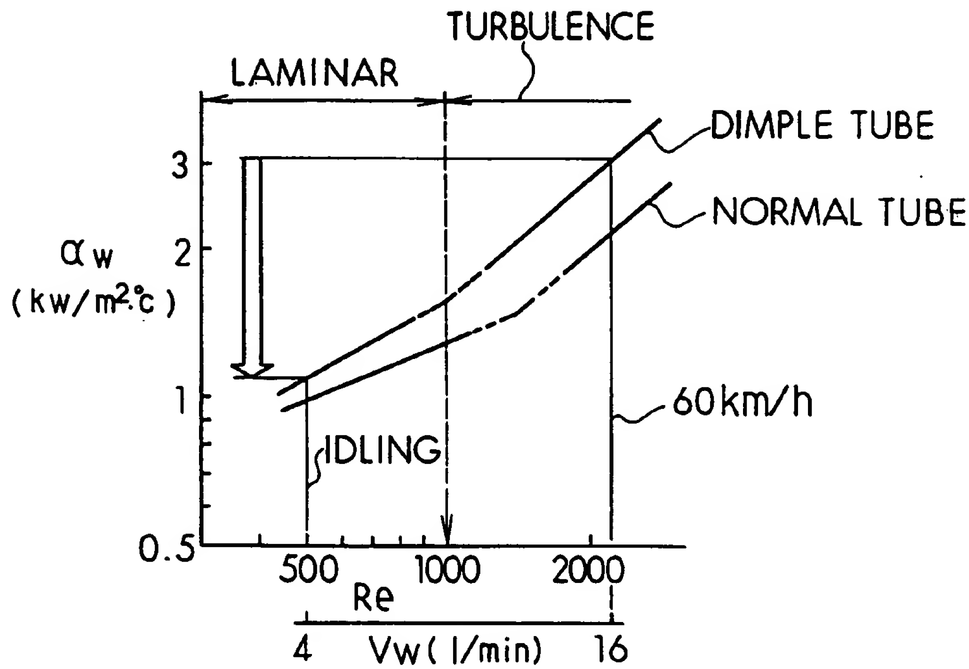

TURBULENCE
LAMINAR
DIMPLE TUBE
NORMAL TUBE
3
2
1
0.5
αw
(kw/m2·°c)
60km/h
IDLING
500
1000
2000
Re
4
Vw( l/min)
16

FIG. 6

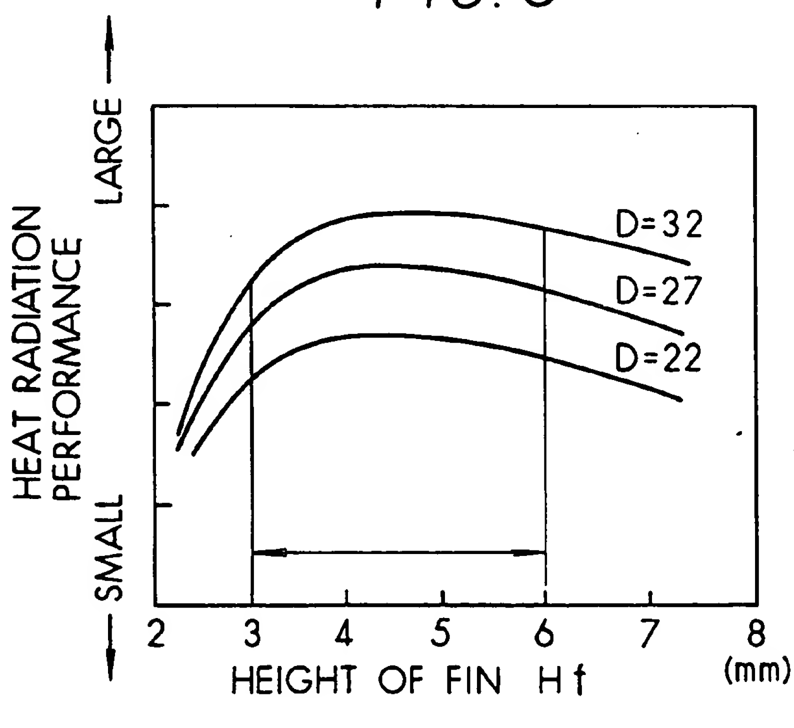
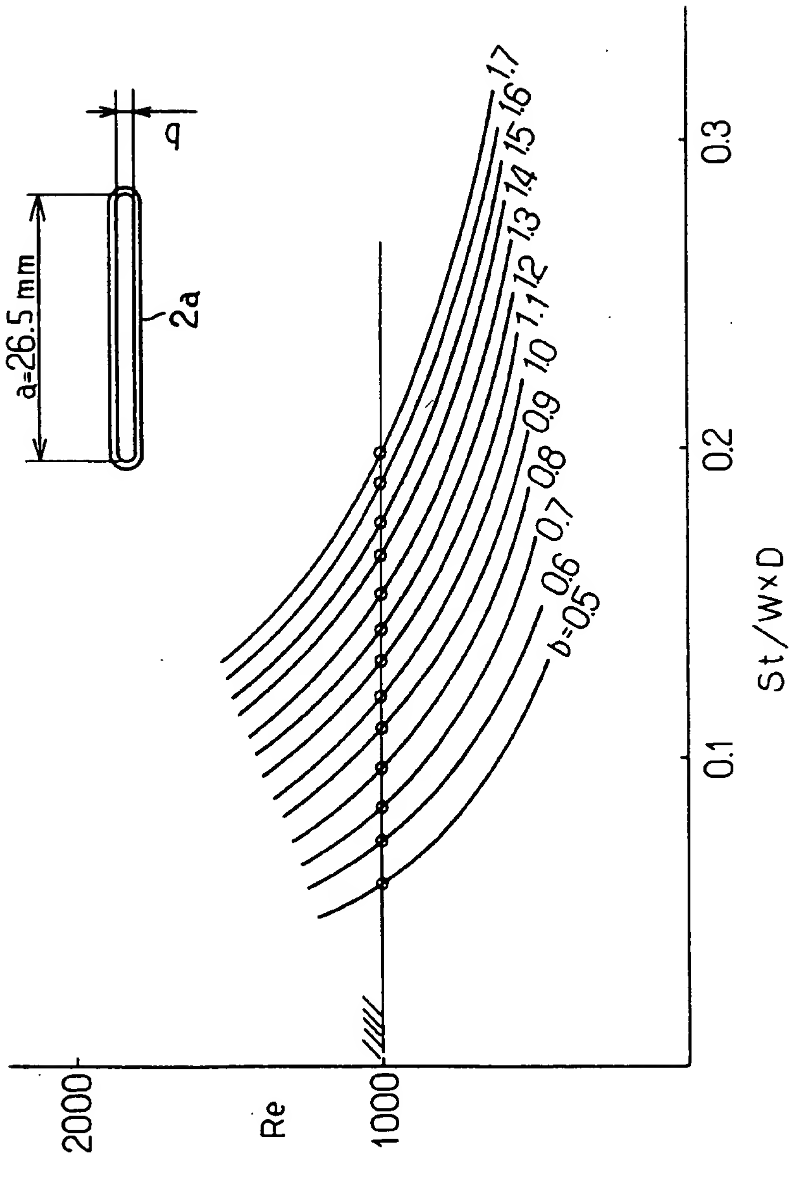
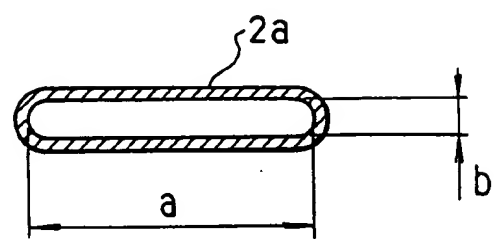

LARGE
SMALL
HEAT RADIATION PERFORMANCE
D=32
D=27
D=22
2
3
4
5
6
7
8
(mm)
HEIGHT OF FIN Hf

FIG. 7


a=26.5 mm
b
2a
Re
2000
1000
0.1
0.2
0.3
St/W×D
b=0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7

F I G. 8


2a
a
b

F I G. 9

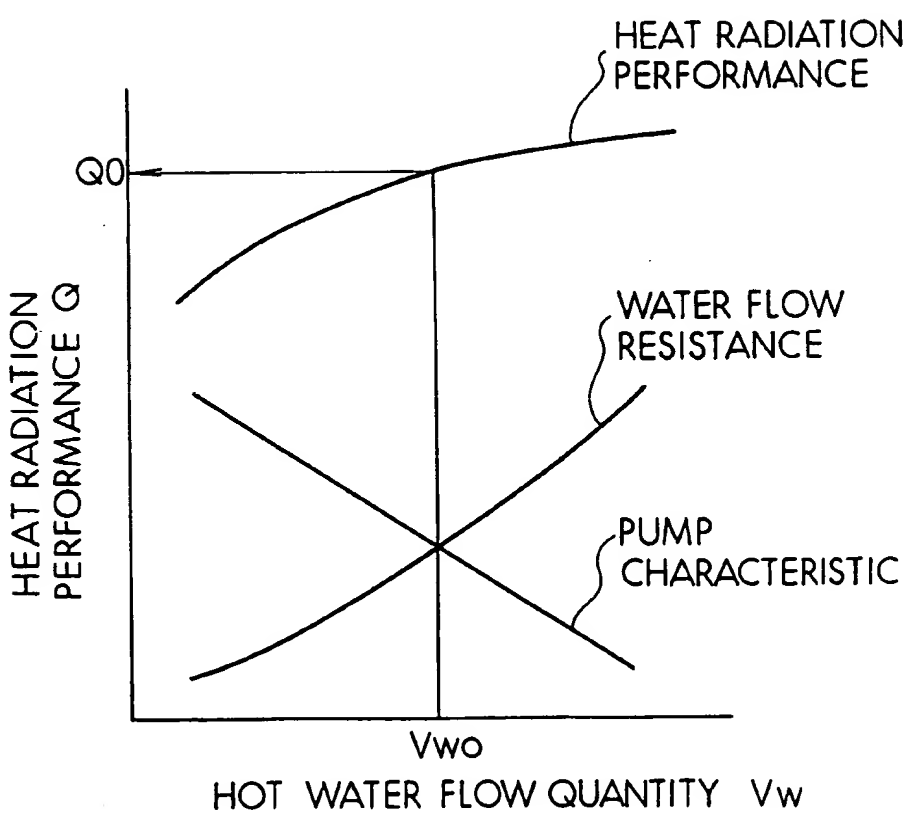

HEAT RADIATION PERFORMANCE
Q0
WATER FLOW RESISTANCE
PUMP CHARACTERISTIC
HEAT RADIATION PERFORMANCE Q
Vwo
HOT WATER FLOW QUANTITY Vw

FIG. 10A

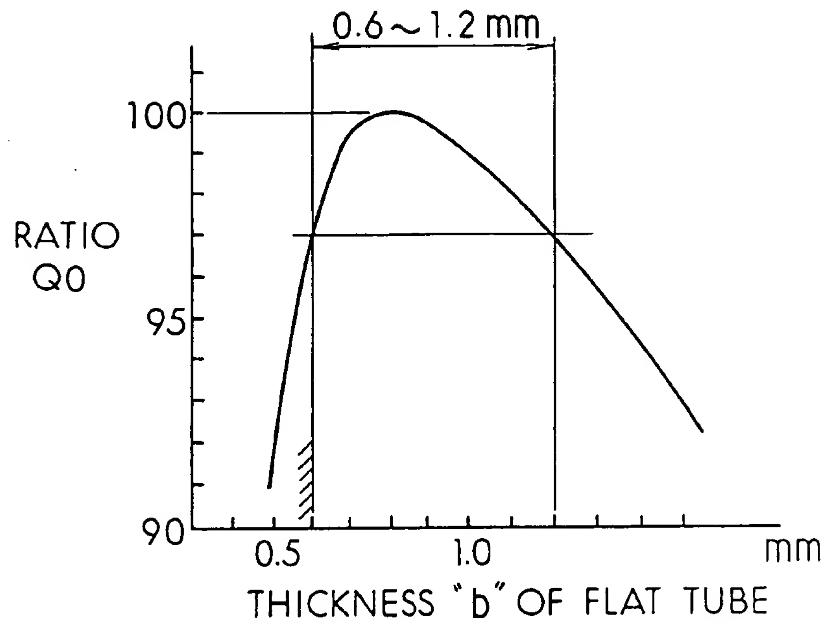



FIG. 10B

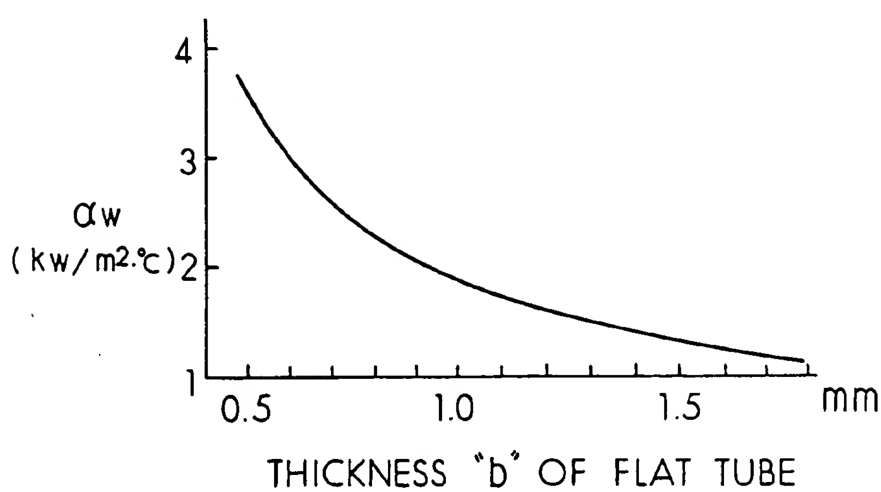
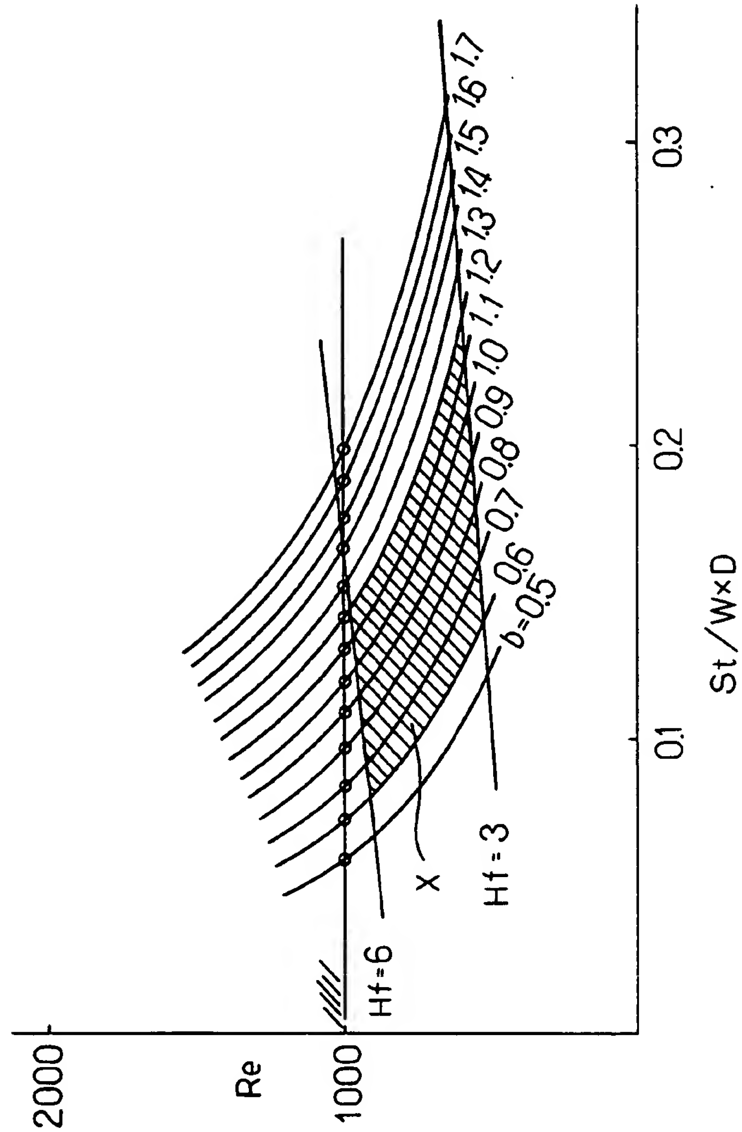
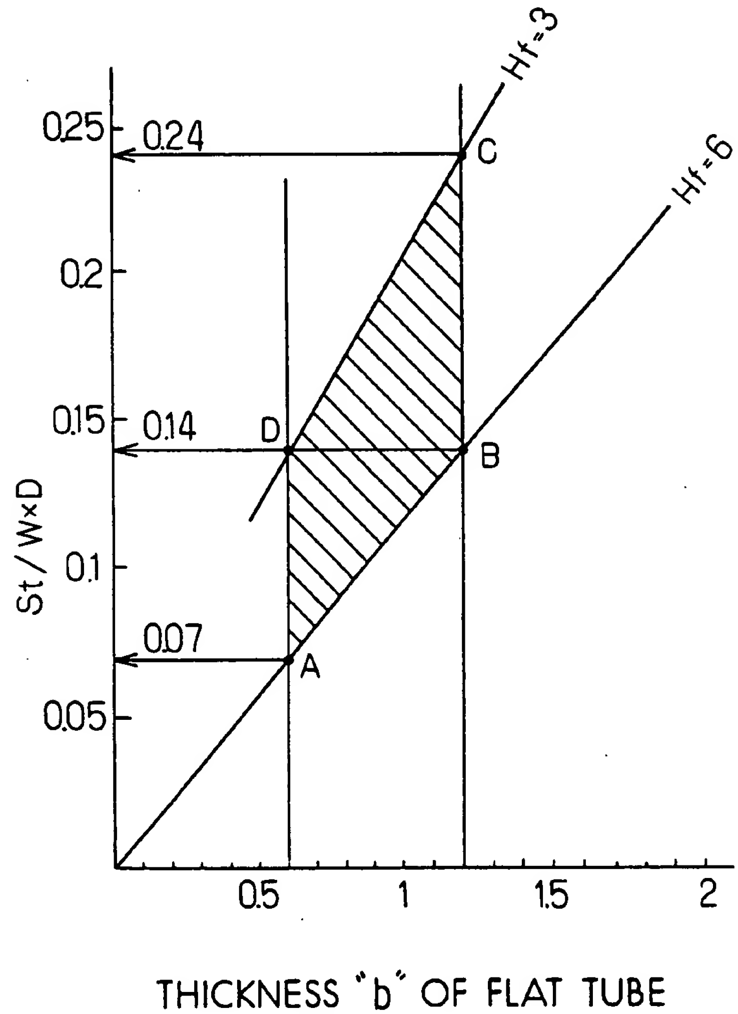


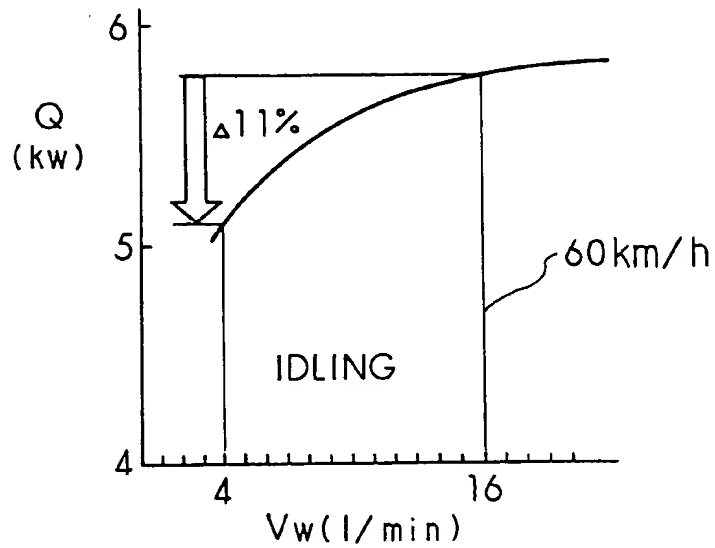
FIG. 11


Re
2000
1000
St/W×D
0.1
0.2
0.3
Hf=6
Hf=3
X
b=0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7

F I G. 12


Hf=3
Hf=6
0.25
0.24
0.2
0.15
0.14
0.1
0.07
0.05
C
D
B
A
St / W×D
0.5
1
1.5
2
THICKNESS "b" OF FLAT TUBE

F I G. 13


6
5
4
Q
(kw)
△11%
60km/h
IDLING
4
16
Vw(l/min)

F I G. 14

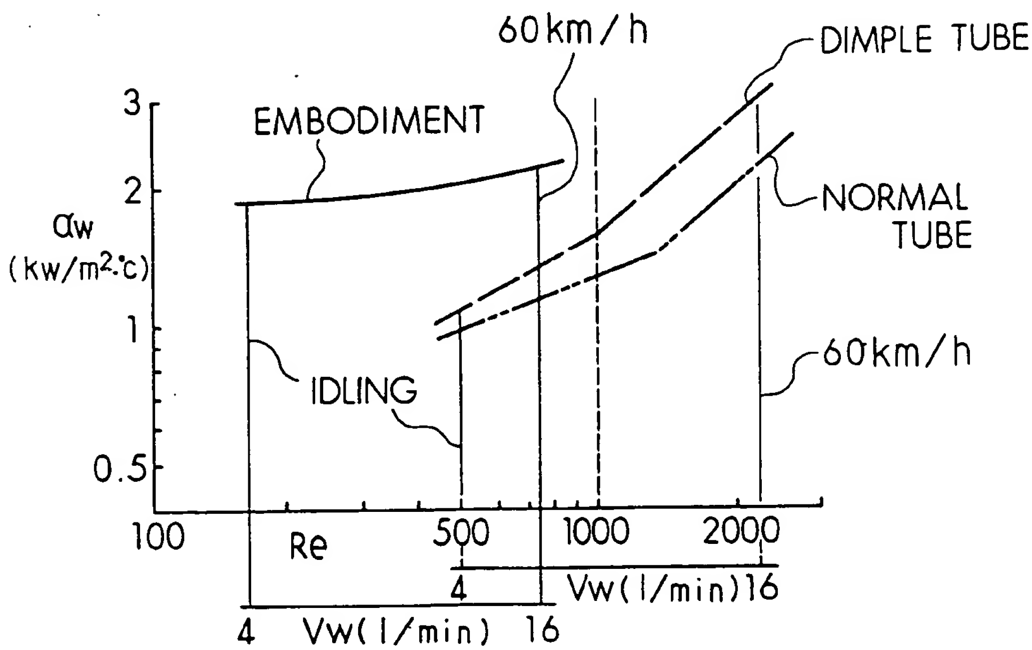
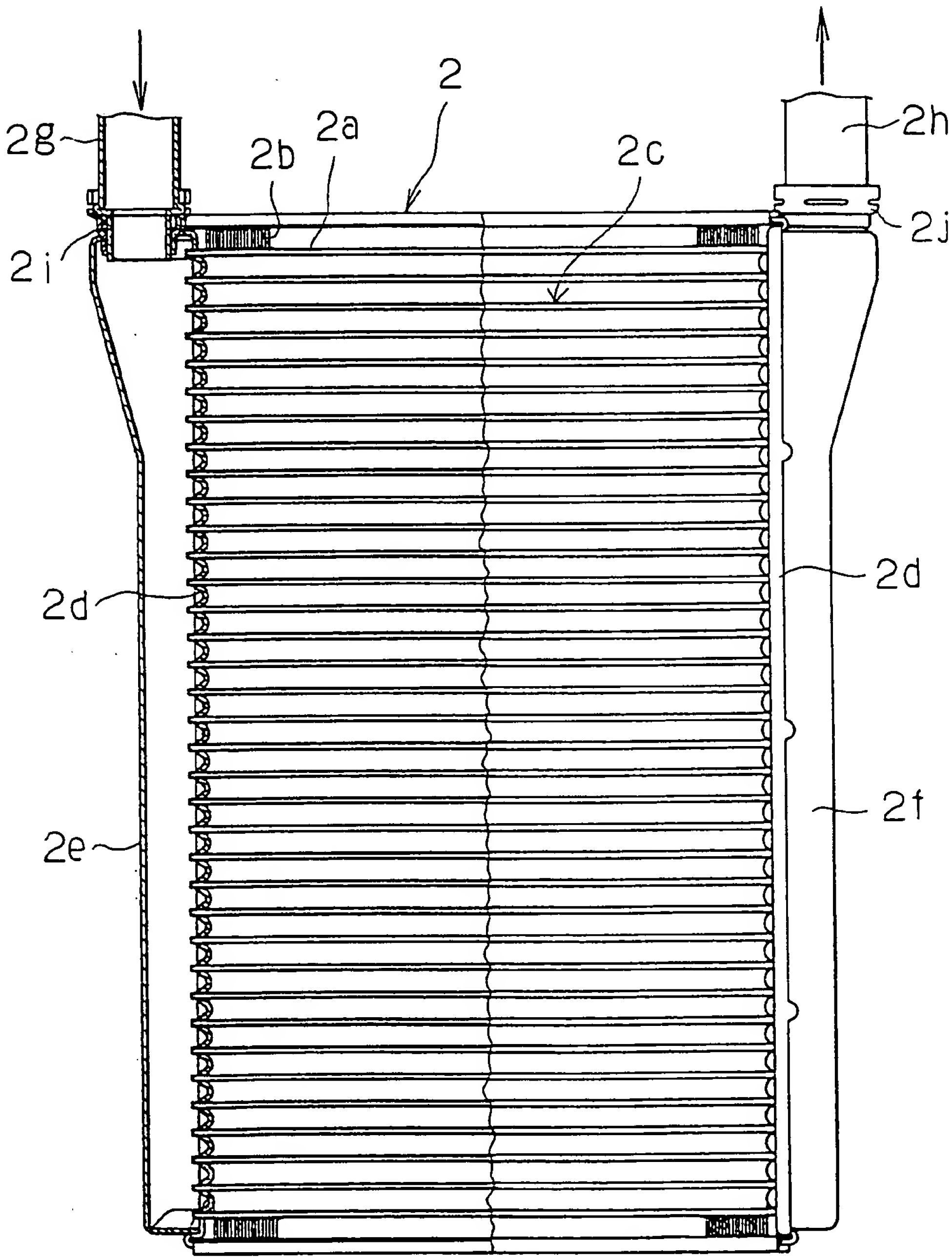

60km/h
DIMPLE TUBE
3
2
1
0.5
αw
(kw/m²·°c)
EMBODIMENT
NORMAL TUBE
60km/h
IDLING
100
Re
500
1000
2000
4
Vw(l/min)16
4 Vw(l/min) 16

FIG. 15



Equivalent to JP 136176

FIG. 16A

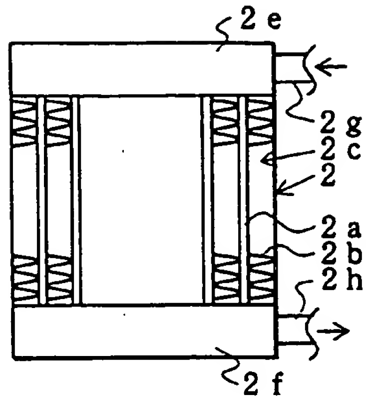



FIG. 16B

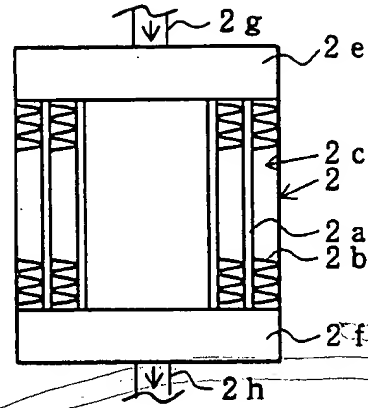



FIG. 16C

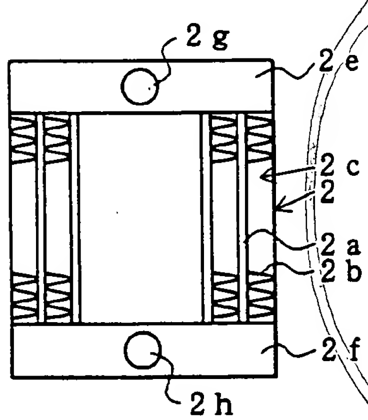



FIG. 16D

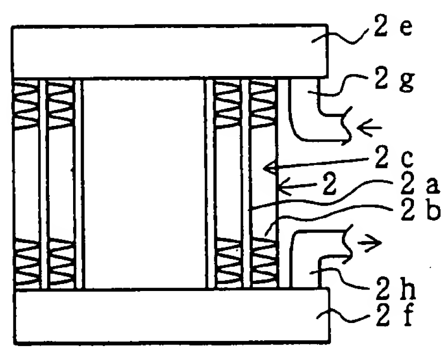



FIG. 16E

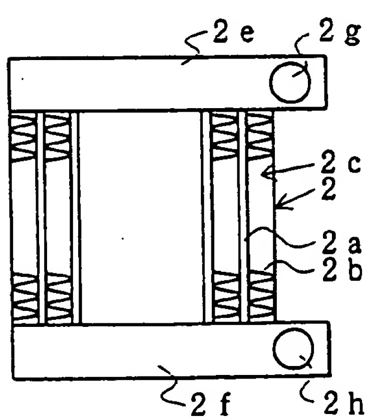



FIG. 16F